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(54) **VARIABLE CAPACITANCE, OBLIQUE SPINDLE TYPE HYDRAULIQUE MACHINE.**

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(73) Proprietor: **HITACHI CONSTRUCTION MACHINERY CO., LTD.**
6-2, Ohtemachi 2-chome
Chiyoda-ku Tokyo 100(JP)

(72) Inventor: **AKASAKA, Yoshimichi**
12-16, Kandatsu Chuuoh 5-chome
Tsuchiura-shi
Ibaraki 300(JP)
Inventor: **NAKAMURA, Ichiro**
2897-33, Mawatari
Katsuta-shi
Ibaraki 312(JP)
Inventor: **GOTOH, Yasuharu**
978-108, Shiratori-cho
Tsuchiura-shi
Ibaraki 300(JP)

(74) Representative: **Patentanwälte Beetz - Timpe - Siegfried Schmitt-Fumian - Mayr**
Steinsdorfstrasse 10
D-80538 München (DE)

EP 0 389 644 B1

EP 0 389 644 B1

Description

This invention relates to a bent axis type variable displacement hydraulic machine which is suitable for use as a bent axis type hydraulic pump, hydraulic motor or the like, and which is adapted to support a rotational shaft by partial-and/or total-hydrostatic bearings according to the first part of claim 1 or claim 8.

Generally, a bent axis type hydraulic machine has the drive disc of a rotational shaft coupled with a cylinder block through pistons which are reciprocably received in the cylinder block. Therefore, in a case where the bent axis type hydraulic machine is applied as a hydraulic pump, the hydraulic reaction forces which act on pistons on the high pressure side in the discharge stroke are supported by the rotational shaft through the drive disc. Similarly, when applied as a hydraulic motor, the hydraulic reaction forces which act on pistons on the high pressure side in the suction (feeding) stroke are supported by the rotational shaft through the drive disc.

Accordingly, in a bent axis type hydraulic machine of this sort, the rotational shaft is subject to radial and thrust load of the hydraulic reaction forces and therefore it is necessary to hold the rotational shaft in suitable condition for supporting these loads.

In this regard, it has been the general practice in the prior art to resort to the so-called mechanical support type which mechanically supports the rotational shaft rotatably by means of ball or roller bearings capable of supporting the radial and thrust loads, the partial hydrostatic support type which mechanically supports either the radial or thrust loads by a roller or ball bearing while supporting the other load hydraulically by a hydrostatic bearing, or the total hydrostatic support type which supports the entire loads hydraulically by hydrostatic bearings.

Of these various shaft supporting means, a hydraulic machine employing a shaft support bearing of the partial hydrostatic type is described, for example, in the JP-A-60-224981, wherein a rotational shaft is supported by a hydrostatic thrust bearing composed of a stationary bearing and a movable bearing, which movable bearing being provided with springs in an outer ring to counteract the thrust load which acts on the rotational shaft, along with pistons which are located on the side of the outer ring to generate a pressure in the same direction as the springs and to which oil pressure is applied from the high pressure area in the cylinder block.

On the other hand, a hydraulic machine supporting a shaft by total-hydrostatic bearings is described in the JP-A-59-131776, which is provided with a radial load bearing sleeve and a thrust load bearing plate within a casing, in combination with a drive flange which is provided movably between the bearing sleeve and the bearing plate to serve also as a drive disc. The drive flange one end face thereof securely connected to a rotational shaft and the other end face coupled with pistons. Further, pressure chambers which constitute a hydrostatic radial bearing is defined the outer peripheral surface of the drive flange and the bearing sleeve, and drive shoes which constitute a hydrostatic thrust bearing are provided on one end face of the drive flange. The afore-mentioned pistons have oil passages bored therein for supplying high pressure oil to the radial and thrust bearings from cylinders in the cylinder block, thereby to hydrostatically support the radial and thrust loads.

In this connection, even if the hydraulic reaction forces which are applied to the rotational shaft through pistons (hereinafter referred to simply as "hydraulic reaction force" for brevity) are same, the resulting radial and thrust loads vary depending upon the tilt angle of the cylinder block. More specifically, the radial load F_R and thrust load F_T are expressed as

$$\left.\begin{aligned} F_R &= F \sin \theta \\ F_T &= F \cos \theta \end{aligned}\right\} \cdots\cdots\cdots\cdots (1)$$

where F is the hydraulic reaction force by the piston, θ is the angle of inclination or tilt angle, F_R is the radial load, and F_T is the thrust load. When the tilt angle θ is minimum, the radial load F_R becomes minimum while the thrust load F_T becomes maximum. On the other hand, when the tilt angle θ is maximum, the radial load F_R becomes maximum while the thrust load F_T becomes minimum.

In short, the above-described prior art devices are arranged to supply directly the hydrostatic bearing or bearings with high pressure oil of a certain level which is generated in cylinders on the high pressure side of a cylinder block (in case of a pump) or which is fed to cylinders on the high pressure side (in case of a motor).

In this manner, in spite of the fact that the radial and thrust loads of the hydraulic reaction forces vary in synchronism with variations in tilt angle of the cylinder block, the conventional counterparts have been arranged simply to apply a high oil pressure of a certain level to a hydrostatic bearing. It follows that the hydrostatic bearing has constant characteristics in load supporting capacity, more specifically, in statically hydrodynamic and dynamically hydrodynamic loads supporting capacity, forming an oil film of an increased thickness on the guide surface of the hydrostatic bearing when the load of the hydraulic reaction force is of a light one (i.e., when the hydrostatic bearing capacity is higher than the load of the hydraulic reaction force), balancing the hydrostatic bearing capacity with the load of the hydraulic reaction force in supporting the latter.

The support of this sort has a problem that the thickness of the oil film is increased to an excessive degree. In this connection, it is known that the rate of oil leakage from an oil film formed on a given sliding surface is proportional to the cube of the oil film thickness. An oil film which has an excessively large thickness as mentioned hereinbefore involves a greater rate of oil leakage from the hydrostatic bearing guide surface, which will lead to a problem of increased power loss.

On the other hand, in a case where the tilt angle of the cylinder block is changed frequently during operation of a pump or motor, the support capacities of the hydrostatic thrust and radial bearings are varied each time when the cylinder block is tilted. This will be reflected by degradations in accuracy of the drive disc positioning in the radial and thrust directions, increasing vibrations of the hydraulic machine to such a degree as will hinder stable rotational movements in high speed operation and impair the durability of the machine.

The DE-A- 29 32 583 discloses a hydraulic axial piston machine, comprising a casing for receiving a cylinder block and a valve plate. A plurality of pistons reciprocally received in bores of said cylinder block are pivotally supported by a drive disc which is fixed to a rotational shaft. The cylinder block is tilted at a fixed angle to the axis of said rotational shaft. A plurality of hydrostatic radial bearings is provided between the casing and the drive disc and connected to the high pressure side of the machine by fixed throttle means.

From the US-A-4 836 693 it is known a hydraulic axial piston machine comprising an intermittently assisted hydrostatic bearing connected with a pressurized fluid supply source. An additional pressure producing device communicates with the pressure fluid channel of the supply source and provides a greater than 100 % pressure balance in the hydrostatic bearing in response to a command signal. The speed of the rotational shaft of the machine is measured by a speed sensor and the output signal of said sensor is combined with a position command signal for producing a signal for activating the pressure producing device.

The present invention contemplates to solve the above-mentioned problems or drawbacks of the prior art, and has as its object the provision of a bent axis type variable displacement hydraulic machine employing a partial-and/or total-hydrostatic bearing support which can ensure operations with reduced oil leakage and of high stability and reliability even under conditions involving intermittent or continual changes of the tilt angle of the cylinder block.

In accordance with the present invention, the above-mentioned object is achieved by a bent axis type variable displacement hydraulic machine which is characterized in that the variable throttle means positioned between a head casing and a valve plate draw out therethrough a pressure commensurate with the tilt angle of the cylinder block, for supply to at least one of hydrostatic radial and thrust bearings.

According to an aspect of the invention, the above-mentioned variable throttle means is constituted by an oil groove formed on one of a head casing and a valve plate along and in communication with one of paired suction and discharge passages whichever is on the high pressure side of the head casing or along and in communication with one of paired suction and discharge ports whichever is on the high pressure side of the valve plate in such a manner as to become deeper at a larger tilt angle, and an oil hole formed on the other one of the head casing and the valve plate in a position opposing the oil groove, drawing through the oil groove or oil hole on the head casing a bearing control pressure which increases with the tilt angle of the valve plate, and supplying the pressure to the hydrostatic radial bearing.

According to another aspect of the invention, the above-mentioned variable throttle means is constituted by an oil groove formed on one of a head casing and a valve plate along and in communication with one of paired suction and discharge passages whichever is on the high pressure side of the head casing or along and in communication with one of paired suction and discharge ports whichever is on the high pressure side of the valve plate in such a manner as to become shallower at a larger tilt angle, and an oil hole formed on the other one of the head casing and valve plate in a position opposing the oil groove, drawing out through the oil groove or oil hole on the head casing a bearing control pressure which decrease with the angle of inclination of the valve plate, and supplying the pressure to the hydrostatic thrust bearing.

Further, according to the invention, the bent axis type variable displacement hydraulic machine is characterized by the provision of sensor for detecting the tilt angle of a cylinder block and a valve plate established by a tilting mechanism, an oil passage for drawing out a pressure from either one of paired suction and discharge passages whichever is on the high pressure side of a head casing and supplying the pressure to at least one of radial and thrust hydrostatic bearings, and a control valve provided within the length of the oil passage for modulating the pressure on the basis of a signal of tilt angle received from the sensor.

The bent axis type variable displacement hydraulic machine of the present invention is applicable as a pump of main hydraulic pressure source in hydraulic systems for construction machines, screw down mechanisms of rolling mill, sea water hydraulic systems and the like.

With the above-described construction, the discharge pressure of a hydraulic pump or the output pressure of a hydraulic motor is supplied to a hydrostatic bearing through the variable throttle means or control valve which modulates the pressure into a bearing control pressure commensurate with the tilt angle.

Consequently, in case the hydrostatic bearing is a radial bearing, there is generated a bearing control pressure which becomes higher with an increase in tilt angle of the cylinder block thereby to support the radial load which is exerted on the drive disc by the hydraulic reaction force. On the other hand, in case the hydrostatic bearing is a thrust bearing, there is generated a bearing control pressure which becomes lower with an increase in tilt angle of the cylinder block thereby to support the thrust load which acts on the drive disc.

Thus, it becomes possible to support stably the radial and/or thrust load which varies with the angle of inclination of the cylinder block, preventing instable vibrations of the rotational shaft and drive disc and maintaining the positioning accuracy of the drive disc in the radial and thrust directions irrespective of variations in the tilt angle, while permitting to reduce the rate of oil leakage from the hydrostatic bearing guide surface.

BRIEF DESCRIPTION OF THE DRAWINGS

In the accompanying drawings:

Figs. 1 through 9 illustrate a first embodiment of the invention, of which Fig. 1 is a longitudinal section of a hydraulic pump embodying the invention; Fig. 2 is an enlarged sectional view of major components of the pump of Fig. 1; Fig. 3 is a front view of a valve plate, taken from the side of a sliding surface on a head cassing; Fig. 4 is a sectional view taken on line IV-IV of Fig. 3; Fig. 5 is a similar sectional view taken on line V-V of Fig. 3; Fig. 6 is a front view of the head casing, taken from the side of a tilting sliding surface; Fig. 7 is a sectional view taken on line VII-VII of Fig. 6; Fig. 8 is a diagram showing the relationship of the radial bearing oil groove depth and bearing control pressure with the tilt angle; and Fig. 9 is a diagram showing the relationship of the thrust bearing oil groove depth and bearing control pressure with the tilt angle;

Figs. 10 to 14 illustrate a second embodiment of the invention, of which Fig. 10 is a front view of a valve plate, taken from the side of a sliding surface on a head casing; Fig. 11 is a sectional view taken on line XI-XI of Fig. 10; Fig. 12 is a front view of the head casing, taken from the side of a tilting sliding surface; Fig. 13 is a sectional view taken on line XIII-XIII of Fig. 12; and Fig. 14 is a sectional view taken on line XIV-XIV of Fig. 12;

Fig. 15 is a longitudinal section of a hydraulic pump in a third embodiment of the invention;

Fig. 16 is a longitudinal section of a hydraulic pump in a fourth embodiment of the invention;

Fig. 17 is a hydraulic circuit diagram of a hydraulic system of a construction machine, to which the present invention is applied;

Fig. 18 is a sectional view of a hydraulic screw down mechanism of rolling mill, to which the present invention is applied; and

Fig. 19 is a diagrammatic view of a sea water hydraulic system.

BEST MODE FOR CARRYING OUT THE INVENTION

Hereinafter, the invention is described more particularly by way of variable displacement type hydraulic pumps shown in the drawings as preferred embodiments.

Referring first to Figs. 1 through 9 showing a first embodiment of the invention, indicated at 1 is a casing which is constituted by a casing body 2 with a bearing portion 2A of a smaller diameter and a tilted cylindrical portion 2B of a larger diameter, and a head casing 3 closing the open outer end of the tilted

cylindrical portion 2B of the casing body 2.

Denoted at 4 is a bearing sleeve which is provided in the bearing portion 2A of the casing 1, and constituted by a sleeve portion 4A fitted in the bearing portion 2A, and a flange portion 4B abutted against a stepped wall portion 2C of the tilted cylindrical portion 2B. Inserted into the bearing sleeve 4 from outside of the casing 1 is a rotational shaft 5 which has a drive disc 6 of a larger diameter integrally formed at its inserted inner end which is extended into the tilted cylindrical portion 2B. The rotational shaft 5 is journalled in the bearing sleeve 4 through a bearing 7, while the drive disc 6 is arranged to support hydraulic reaction forces through a hydrostatic radial bearing 23 and a hydrostatic thrust bearing 28 as will be described hereinafter.

The reference numeral 8 designates a cylinder block which is provided within the casing 1 and rotatable integrally with the rotational shaft 5, the cylinder block 8 having a plural number of pistons 10 reciprocably received in cylinders 9 which are bored axially in the cylinder block 8. The pistons 10 are each provided with a spherical portion 10A at the fore end and thereby pivotally connected with the drive disc 6.

Indicated at 11 is a square valve plate which forms, on one side thereof, a flat plate-like switching surface 11A for sliding contact with the opposing end face of the cylinder block 8 and, on the other side, a convex sliding surface 11B for sliding contact with a concave tilting sliding surface 15 which is formed on the head casing 3 as will be described hereinafter. A pair of suction and discharge ports, namely, a suction port 12 and a discharge port 13 are bored in the valve plate 11. These ports 12 and 13 form, on the side of the switching surface 11A, an arcuate suction port opening 12A and an arcuate discharge port opening 13A, respectively, which are intermittently communicated with each one of the cylinders 9 by rotation of the cylinder block 8, while respectively forming a rectangular suction port opening 12B and a slot-like discharge port opening 13B on the side of the sliding surface 11B. (Fig. 3)

Designated at 14 is a center shaft which tiltably supports the cylinder block 8 between the drive disc 6 and the valve plate 11, and has at one end a spherical portion 14A which is pivotally supported in a center position of the drive disc 6. The other end of the center shaft 14, which is protruded through the cylinder block 8, is slidably received in a through hole 11C which is bored at the center position of the valve plate 11 for centering the cylinder block 8 and the valve plate 11.

On the other hand, the head casing 3 is provided with a concave arcuate tilting sliding surface 15 on its inner wall surface, the tilting slide surface 15 having fluid-tight seal lands 15A and 15B which are in sliding contact with the sliding surface 11B of the valve plate 11. The head casing 3 is provided with a pair of suction and discharge passages, namely, a suction passage 16 and a discharge passage 17, the suction passage 16 opening into a recess between the seal lands 15A and 15B on the tilting slide surface 15 for communication with the suction port 12 of the valve plate 11 while the discharge passage 17 opening onto the seal land 15B for communication with the discharge port 13. (See Fig. 6.)

The reference numeral 18 denotes a tilting mechanism which is mounted in the head casing 3 for tilting the valve plate 11 along the tilting slide surface 15. The tilting mechanism 18 is constituted by a cylinder bore 19 formed in the head casing 3 and having oil passages 19A and 19B at the opposite ends thereof, a servo piston 21 slidably fitted in the cylinder bore 19 and defining oil chambers 20A and 20B on the outer sides of its opposite axial ends, and a rocking pin 22 fitted in the servo piston 21 and having a spherical portion at its distal end pivotally fitted in the valve plate 11. The oil pressure which is received from an auxiliary pump (not shown) through a tilting control valve is supplied to the oil chamber 20A or 20B through the oil passage 19A or 19B, thereby driving the servo piston 21 to tilt the valve plate 11 and cylinder block 8.

The hydrostatic bearings employed in this embodiment have the constructions as described below.

Firstly, indicated at 23 is a hydrostatic radial bearing for supporting the radial load components of the hydraulic reaction forces which are exerted on the drive disc 6 by pistons. The hydrostatic radial bearing 23 is constituted by: a bearing sleeve 24 of a ring-like form located around the drive disc 6 and having the outer periphery thereof fitted in the inclined cylindrical portion 2B of the casing body 2 and the inner periphery disposed in sliding contact with the outer periphery (the hydrostatic bearing guide surface) of the drive disc 6; a plural number of pressure chambers 25 (at least three or counting in maximum a number equivalent to the number of the pistons 10) in the form of recesses formed at equidistant positions around the inner periphery of the bearing sleeve 24; supply ports 26 formed on the outer periphery of the bearing sleeve 24 at positions corresponding to the pressure chambers 25; and throttle passages 27 provided between the supply ports 26 and the pressure chambers 25 to control the static pressure in the pressure chambers 25 according to the load condition. The respective supply ports 26 of the hydrostatic radial bearing 23 receive the bearing control pressure, which is increased with the tilt angle as will be described hereinafter, for supporting the radial loads. The supply ports 26 may be substituted by a single annular groove if desired.

On the other hand, denoted at 28 is a hydrostatic thrust bearing for supporting the thrust load components of the hydraulic reaction forces which are exerted on the drive disc 6 through the pistons. The hydrostatic thrust bearing 28 is constituted by: the afore-mentioned bearing sleeve 4; a plural number of axial pad insert holes 29 provided at predetermined intervals around the circumference of a flange portion 4B of the bearing sleeve 4; bearing pads 30 each having a pad portion 30A and a shaft portion 30B smaller than the pad portion in diameter, the pad portion 30A being held in sliding contact with the back surface (the hydrostatic bearing guide surface) of the drive disc 6, and the shaft portion B being inserted in a corresponding one of the pad insert holes 29; pressure chambers 31 in the form of sectionally U-shaped grooves formed on the faces of the respective pad portions 30A on the side of the sliding contact surface of the drive disc 6; supply chambers 32 defined in the pad insert holes 29 by the shaft portions 30B; and throttle passages 33 intercommunicating the supply chambers 32 and the pressure chambers 31. Each supply chamber 32 of the hydrostatic thrust bearing 28 receives the bearing control pressure, which is lowered with the tilt angle as will be described hereinafter, for supporting the thrust loads. It suffices to provide the bearing pads 30 at a number of spaced positions around the flange portion 4B of the bearing sleeve 4.

Described below is the construction of the variable throttle mechanism which produces the bearing control pressure varying commensurate with the tilt angle of the cylinder block 8.

Namely, the variable throttle mechanism which is generally indicated at 34 produces the bearing control pressure for the radial hydrostatic bearing 23, and includes: an oil groove 35 formed on the sliding surface 11B of the valve plate 11 at a position in the vicinity of and along one side of the discharge opening 13B of the discharge port 13 from a median point toward the lower end of the latter; and an oil hole 36 formed in the seal land 15B in a position at one side of the opening of a discharge passage 17 on the side of the tilting slide surface 15 of the head casing 3 and opposingly to the afore-mentioned oil groove 35.

In this instance, as shown particularly in Figs. 4 and 8, the oil groove 35 is formed in a wedge-like shape having a continuously varying depth h which becomes smallest when the tilt angle θ of the valve plate 11 is zero or when $\theta = 0^\circ$ (with the valve plate 11 in the uppermost position in Fig. 1), and which becomes greatest when the tilt angle θ is maximum or when $\theta = \theta$ max (the position of Fig. 1). At the deepest end, the oil groove 35 is provided with a groove portion 35A which communicates with the discharge port opening 13B. The oil passage 36 is formed such that it confronts the lower end (the shallowest end) of the oil groove 35 at the minumum tilt angle, and has an open area which produces a maximum discharge pressure Pdmax at the maximum tilt angle. Thus, the variable throttle mechanism 34 for the radial bearing is capable of producing a bearing control pressure Pd which has characteristics of increasing in proportion to the tilt angle θ. (See Fig. 8.)

Indicated at 37 is a variable throttle mechanism for producing a bearing control pressure for the hydrostatic thrust bearing 28, the variable throttle mechanism 37 including: an oil groove 38 formed on the sliding contact surface 11B of the valve plate 11 in the vicinity of and along the other side of the discharge port opening 13B opposingly to the above-mentioned oil groove 35; and an oil hole 39 formed in the seal land 15B on the tilting slide surface 15 of the head casing 3 in a position confronting the oil groove 38 and on one side of the discharge passage 17 away from the oil hole 36.

In this instance, as shown particularly in Figs. 5 and 9, the oil groove 38 is formed in a wedge-like shape having a continuously varying depth h which becomes greatest when the tilt angle θ of the valve plate 11 is zero or when $\theta = 0^\circ$ (when the valve plate 11 is in the uppermost position in Fig. 1), and which becomes smallest when the tilt angel θ is maximum or when $\theta = \theta$ max (the position of Fig. 1). At the shallowest end, the oil groove 38 is provided with a groove portion 38A which communicates with the discharge port opening 13B. The oil hole 39 is formed such that it confronts the lower end (the shallowest end) of the oil groove 38 at the minimum tilt angle, and has an open area which produces a maximum discharge pressure Pdmax at the maximum tilt angle. Thus, the variable throttle mechanism 37 for the thrust bearing is arranged to produce a bearing control pressure Pd with characteristics of becoming lower in inverse proportion to the tilt angle θ. (See Fig. 9.)

Further, designated at 40 is an oil passage for the radial bearing control pressure, which is bored into a thick wall portion of the casing 1 and has one end thereof in communication with the oil hole 36 and the other end in communication with the respective supply ports of the radial hydrostatic bearing 23. On the other hand, indicated at 41 is an oil passage for the thrust bearing control pressure, which is bored into a thick wall portion of the casing 1 and has one end thereof in communication with the oil hole 39 and the other end in communication with the respective supply chambers of the trust hydrostatic bearing 28.

When applied as a hydraulic pump, the bent axis type hydraulic machine of this embodiment operates in the manner as follows.

Firstly, the valve plate 11 is tilted to the maximum tilted position of Fig. 1 together with the cylinder block 8 by operation of the tilting mechanism 18. For this purpose, the servo piston 21 is displaced by supplying the oil pressure from the auxiliary pump to the oil chamber 20A of the cylinder 19. By so doing, the pivoting pin 22 is displaced together with the servo piston 21, tilting the valve plate 11 under guidance of the tilting slide surface 15. Consequently, the cylinder block 8 is tilted integrally with the center shaft 14 into the position shown in Fig. 1, with its rotational axis inclined relative to the axis of the rotational shaft 5.

Nextly, the rotational shaft 5 is rotated by an engine, electric motor or other suitable drive source, whereupon the cylinder block 8 is rotated integrally with the rotational shaft 5 since the drive disc 6 of the rotational shaft 5 is connected to the pistons 10 in the respective cylinders 9 of the cylinder block 8. Consequently, the pistons 10 are reciprocated in the respective cylinders 9 during rotation of the cylinder block 8. During the suction stroke when each piston 10 is moved away from the cylinder 9, the operating oil is drawn into the cylinder 9 through the suction port 12 and suction passage 16, and, during the discharge stroke when the piston 10 is moved into the cylinder 9, the operating oil in the cylinder 9 is pressurized and discharged through the discharge port 13 and discharge passage 17.

In this connection, in a bent axis type hydraulic pump of this sort, in proportion to the number of pistons for generating the discharge pressure (e.g., in case the total number of pistons is seven, the maximum number of the pressurizing pistons is four, the minimum number of the pressurizing pistons is three, and the average number of the pressurizing pistons is 3.5), the load of the hydraulic reaction force of the piston and the moment load are exerted on the drive disc 6 in synchronism with the rotational speed of the rotational shaft 5. As shown particularly in Fig. 2, the load F exerted on the drive disc 6 is divided at the support surface of the spherical portion 10A of the piston rod 10 into a radial load or a radial component F_R and a thrust load or an axial component F_T according to the tilt angle θ. The load consisting of components of two different directions and the moment load are supported by the hydrostatic radial and thrust bearings 23 and 28. Namely, the loads are supported in the radial and axial directions by the hydrostatic and hydrodynamic sliding bearing actions of the static pressures in the pressure chambers 25 and 31 of the hydrostatic bearings 23 and 28.

Now, studying more closely the loads which are exerted on the drive disc 6 in the radial and axial directions, the load F resulting from the hydraulic reaction forces varies depending upon the number of pistons and at the same time varies with the tilt angle θ as expressed by Equations (1) hereinbefore. More specifically, the radial load F_R becomes smallest when the tilt angle θ is minimum and becomes greatest when the tilt angle θ is maximum. On the other hand, the thrust load F_T becomes greatest when the tilt angel θ is minimum, and becomes smallest when the tilt angel θ is maximum.

Therefore, according to the present embodiment, the radial bearing control pressure to be supplied to the hydrostatic radial bearing 23 is produced through the variable throttle mechanism 34 for the radial bearing. More specifically, the variable throttle mechanism 34 for the radial bearing employs, on the part of the valve plate 11, the oil groove 35 which becomes deeper at a larger tilt angle θ and which is in communication with the discharge port 13, and, on the part of the head casing 3, the oil hole 36 which is constantly in communication with the oil groove 35, producing from the oil hole 36 the bearing control pressure Pd which becomes higher at a larger tilt angle θ, for supply to the hydrostatic radial bearing 23 through the oil passage 40.

Accordingly, when the tilt angle θ of the cylinder block 8 is intermittently or continuously varied from the minimum tilt angle ($\theta = 0°$) to the maximum tilt angle ($\theta = \theta$ max), the bearing control pressure Pd is also elevated in synchronism with the tilt angle θ. (See Fig. 8.) As a result, the static pressure which prevails in the pressure chamber 25 of the hydrostatic radial bearing 24 is elevated correspondingly to the bearing control pressure Pd to securely support the radial load F_R which increases with the tilt angle θ.

On the other hand, the bearing control pressure to be supplied to the hydrostatic thrust bearing 28 is produced by the variable throttle mechanism 37 for the thrust bearing. More specifically, the variable throttle mechanism 37 for the thrust bearing employs, on the part of the valve plate 11, the oil groove 38 which becomes shallower at a larger tilt angle θ and which is in communication with the discharge port 13, and, on the part of the head casing 3, the oil hole 39 which is constantly in communication with the oil groove 38, producing from the oil hole 39 the bearing control pressure Pd which becomes lower at a larger tilt angle θ, for supply to the hydrostatic thrust bearing 28 through the oil passage 41.

Accordingly, when the tilt angle θ of the cylinder block 8 is intermittently or continuously varied from the minimum tilt angle ($\theta = 0°$) to the maximum tilt angle ($\theta = \theta$ max), the bearing control pressure Pd is also lowered in synchronism with the tilt angle θ. (See Fig. 9.) As a result, the static pressure which prevails in the pressure chamber 31 of the hydrostatic thrust bearing 28 is lowered correspondingly to the bearing control pressure Pd to securely support the thrust load F_T which increases with the tilt angle θ.

Thus, according to the present invention, the depths h of the oil grooves 35 and 38 are automatically determined in correspondence to the tilt angle θ of the cylinder block 8, so that it is possible to produce the bearing control pressures Pd which corresponds to the respective oil groove depths h. Consequently, the drive disc 6 can be supported stably irrespective of variations in the radial and thrust loads which are imposed on the drive disc 6 by the hydraulic reaction forces according to the tilt angle θ, while holding the oil leaks from the hydrostatic bearings 23 and 28 to a minimum and reducing vibrations of the rotational shaft 5.

Referring now to Figs. 10 to 14, there is shown a second embodiment of the invention, in which the component parts common to the foregoing first embodiment are designated by common reference numerals and their description is omitted to avoid unnecessary repetitions.

A feature of this embodiment resides in that an oil groove serving as a variable throttle mechanism is provided on the part of the head casing and an oil hole in communication with the discharge port is provided on the part of the valve plate.

More specifically, indicated at 51 is a variable throttle mechanism for the radial bearing, the variable throttle mechanism 51 being located on the side of the tilting slide surface 15 of the head casing 3, and constituted by; an oil groove 52 formed along one side of the discharge passage 17, which is opened on the seal land 15B, and extended downwardly in the tilting direction from a median point of the seal land 15B; and an oil hole 53 formed into the sliding contact surface 11B of the valve plate 11 at one side of the discharge port opening 13B opposingly to the oil groove 52.

As shown in Fig. 13, the oil groove 52 is in the form of a wedge-shaped groove having a depth h which becomes smallest at the minimum tilt angle (θ = O°) of the valve plate and becomes greatest at the maximum tilt angle (θ = θ max), and in communication with the oil passage 40 which is opened at the deepest end of the oil groove 52. On the other hand, as shown in Fig. 11, the oil hole 53 is provided with a communicating passage 53A which is opened into the side wall of the discharge port opening 13B and supplied with part of the discharge pressure.

Thus, as the valve plate 11 is tilted along the tilting slide surface 15 together with the cylinder block 8, the depth h of the oil groove 52 which confronts the oil hole 53 becomes greater correspondingly to the tilt angle θ. Accordingly, the variable throttle mechanism 51 for the radial bearing is capable of producing a bearing control pressure Pd which has characteristic of increasing in proportion to the tilt angle θ in the same manner as in the first embodiment.

Further, designated at 54 is a variable throttle mechanism for the thrust bearing, the variable throttle mechanism 54 being located on the side of the tilting slide surface 15 of the head casing 3, and constituted by; an oil groove 55 formed along the other side of the discharge passage 17, which is opened on the seal land 15B, and extended opposingly to the oil groove 52 and downwardly in the tilting direction from a median point of the seal land 15B; and an oil hole 56 formed into the sliding contact surface 11B of the valve plate 11 opposingly to the oil groove 55 at the other side of the discharge port opening 13B away from the oil groove 52.

As shown in Fig. 14, the oil groove 55 is in the form of a wedge-shaped groove having a depth h which becomes greatest at the minimum tilt angle (θ = O°) of the valve plate 11 and becomes smallest at the maximum tilt angle (θ = θ max), and in communication with one end of the oil passage 41 which is opened at the deepest end of the oil groove 55. On the other hand, as shown in Fig. 11, the oil hole 56 is provided with a communicating passage 56A which is opened into the side wall of the discharge port opening 13B and supplied with part of the discharge pressure.

Thus, as the valve plate 11 is tilted along the tilting slide surface 15 together with the cylinder block 8, the depth h of the oil groove 55 which confronts the oil hole 56 becomes shallower correspondingly to the tilt angle θ. Accordingly, the variable throttle mechanism 54 for the thrust bearing is arranged to produce a bearing control pressure Pd with characteristics of becoming lower in inverse proportion to the tilt angle θ in the same manner as in the first embodiment.

With the above-described arrangement, as the cylinder block 8 is tilted, this embodiment can also produce the bearing control pressure Pd of Fig. 8 by the variable throttle mechanism 51 for the radial bearing and the bearing control pressure Pd of Fig. 9 by the variable throttle mechanism 54 for the thrust bearing to give the same effects as in the first embodiment.

Referring to Fig. 15, there is illustrated a third embodiment of the invention, in which the component parts common to the first embodiment are likewise designated by common reference numerals, and their description is omitted to avoid repetitions.

In the first embodiment described hereinbefore, arrangements are made such that the bearing control pressures produced by the variable throttle mechanisms 34 for the radial bearing and the variable throttle mechanism 37 for the thrust bearing are fed to the supply port 26 of the hydrostatic radial bearing 23 and

the supply chamber of the hydrostatic thrust bearing 28, respectively, through the oil passages 40 and 41 which are formed in the thick wall portion of the casing 1.

However, this embodiment employs external conduits 61 and 62 which are located on the outer side of the casing 1 and extended for connection between the variable throttle mechanism 34 (53) and hydrostatic radial bearing 23 and between the variable throttle mechanism 37 (54) and hydrostatic thrust bearing 28, respectively.

The third embodiment with the above arrangements produces the same effects as in the first embodiment.

Referring to Fig. 16, there is illustrated a fourth embodiment of the invention, in which the component parts common to the first embodiment are designated by common reference numerals and their description is omitted to avoid repetitions.

A feature of this embodiment resides in the arrangements including means for detecting the tilt angle of the cylinder block by the tilting mechanism, and modulating the discharge pressure of the pump (in case of pump operation) or the supply pressure of the motor (in case of motor operation) into bearing control pressures corresponding to the detected tilt angle for supply to the hydrostatic bearings.

In Fig. 16, denoted at 71 is a tilt angle sensor which is mounted, for example, on the head casing 3 and which is adapted to detect the tilt angle θ of the cylinder block 8 or valve plate 11 by the tilting mechanism 18 to produce a tilt angle signal S. For this purpose, for example, there may be employed as the tilt angle sensor 71 a displacement sensor such as a potentiometer or differential transformer which detect the tilt angle by way of the sliding displacement of the servo piston 21. Otherwise, as the tilt angle sensor 71, there may be employed a displacement sensor which detects the sliding displacement of the valve plate 11 or a rotational displacement sensor which directly detects the rotational angle of the cylinder block 8 or center shaft 14.

The reference numerals 72 and 73 indicate suction and discharge ducts which communicate with the suction and discharge passages 16 and 17, respectively. A shuttle valve 74 is provided between the suction and discharge ducts 72 and 73 to select a higher pressure side.

Indicated at 75 is one external duct which is connected between the shuttle valve 74 and the supply port of the hydrostatic radial bearing 23, and at 76 is another external duct which is connected between the shuttle valve 74 and the supply chamber 32 of the hydrostatic thrust bearing 28. An electromagnetic proportional control valve 77 is provided at a suitable position within the length of the external duct 75, dividing same into an inflow duct 75A and an outflow duct 75B. On the other hand, an electromagnetic proportional reducing valve 78 is provided at a suitable position within the length of the external duct 76, similarly dividing same into an inflow duct 76A and an outflow duct 76B.

In this instance, the electromagnetic proportional control valve 77 is constituted by an electromagnetic servo valve which increases its output pressure in proportion to the amount of signal. For this purpose, the exciting coil of the electromagnetic proportional control valve 77 is connected through an amplifier 79 to the tilt angle sensor 71, which supplies the electromagnetic proportional valve 77 with a tilt angle signal S corresponding to the tilt angle θ of the cylinder block 8 to produce the bearing control pressure proportional to the tilt angle signal S for supply to the hydrostatic radial bearing 23. Namely, the electromagnetic proportional valve 77 serves to modulate the pump discharge pressure from the shuttle valve 74 into the bearing control pressure which corresponds to the tilt angle θ for supply to the hydrostatic radial bearing 23 and has the same characteristics as the control pressure shown in Fig. 8.

Further, in this instance, the afore-mentioned electromagnetic proportional reducing valve 78 is constituted by an electromagnetic servo valve the output pressure of which becomes lower in inverse proportion to increases in amount of its input signal. For this purpose, the electromagnetic proportional reducing valve 78 is connected through an amplifier 80 to the tilt angle sensor 71, which supplies the reducing valve 78 with a tilt angle signal S corresponding to the tilt angle θ of the cylinder block 8 to produce the bearing control pressure varying in inverse proportion to the tilt angle signal S for supply to the hydrostatic thrust bearing 28. Namely, the electromagnetic proportional reducing valve 78 serves to modulate the pump discharge pressure from the shuttle valve 74 into the bearing control pressure which becomes lower correspondingly to the tilt angle θ for supply to the hydrostatic thrust bearing 28 and has the same characteristics as the control pressure shown in Fig. 9.

In this embodiment with the above-described arrangements, as the cylinder block 8 and valve plate 11 are tilted by the tilting mechanism 18, the tilt angle sensor 71 produces a tilt angle signal S corresponding to the tilt angle θ of the cylinder block 8. As a result, for supply to the hydrostatic radial bearing 23, the electromagnetic proportional valve 77 produces bearing control pressure Pd which becomes higher in proportion to the amount of the tilt angle signal S. On the other hand, for supply to the hydrostatic thrust bearing 28, the electromagnetic proportional reducing valve 78 produces bearing control pressure Pd which

becomes lower in inverse proportion to the tilt angle signal S.

Thus, this embodiment is capable of stably supporting the drive disc 6 in the same manner as the first embodiment, irrespective of variations in the radial and thrust loads of hydraulic reaction forces of pistons, which are exerted on the drive disc 6 according to the tilt angle θ, while holding the oil leaks from the hydrostatic bearings 23 and 28 to a minimum.

Referring to Figs. 17 to 19, there are shown various examples applying the variable displacement type hydraulic machine of the invention as a pump.

Fig. 17 shows a circuit diagram of a hydraulic system employing the machine of the invention for a hydraulic construction machine like power shovel.

In Fig. 17, indicated at 101 is an engine serving as a drive source, at 102 and 103 are static pressure-support type hydraulic pumps according to the invention, at 104 are a group of control valves for controlling the flow directions of the hydraulic powers from the pumps 102 and 103, at 105 is a rotating motor, at 106 is a center joint for relaying the powers from the group of control valves 104, at 107 and 108 are travelling hydraulic motors mounted on a lower travelling body, at 109 is a bucket operating hydraulic cylinder, at 110 is an arm operating hydraulic cylinder, at 111 is a boom operating hydraulic cylinder, at 112 to 120 are conduits interconnecting the afore-mentioned hydraulic components or elements.

With the construction machine hydraulic system arranged in the above-described manner, the high fluid pressures, which are discharged from the hydraulic pumps 102 and 103 driven from the engine 101, are fed through the control valves 104 to the rotating hydraulic motor 105 which drives the rotating system, the travelling hydraulic motors 107 and 108 which drive the travelling system, or the hydraulic cylinders 109 to 111 for the boom, arm and bucket, to perform an excavating operation.

In a case where the hydraulic pumps 102 and 103 of the above-described construction machine incorporate the hydraulic machine according to the invention, they can operate as hydraulic pumps of high stability and reliability with less oil leaks even when the tilt angles of the hydraulic pumps 102 and 103 are increased for the purpose of enhancing the travelling and excavating powers for a higher performance. Similar effects can be produced when the invention is applied to the rotating motor 105 or to the travelling hydraulic motors 107 and 108.

Fig. 18 illustrates in section a hydraulic screw down mechanism of rolling mill, incorporating the hydraulic machine according to the present invention.

In Fig. 18, indicated at 201 is a mill housing, at 202 is a back-up roll, at 203 is an intermediate roll, and at 204 is a work roll for directly rolling a work 204 into a predetermined thickness. The reference numeral 206 indicates a screw down cylinder with a piston 206A for controlling the thickness of the work 205. Further, designated at 207 are a pair of displacement meters for detecting the position of the piston of the screw down cylinder 206, and at 208 are a pair of force motor valves which convert an electric signal based on a screw down command into a fluid power for controlling the thickness of the work 205. Indicated at 209 is a static pressure support type hydraulic pump according to the invention.

In the work thickness control system for a hydraulic screw down mechanism of rolling mill, the positions of the screw down cylinders 206 are adjusted by the force motor valves 208 according to a screw down command to control the thickness of a work between the paired upper and lower work rolls 204 with high precision in the order of microns. When the hydraulic pump 209 of the present invention is applied to a hydraulic system of this sort, it becomes possible to obtain the same effects as explained hereinbefore.

Further, Fig. 19 diagrammatically shows a sea water hydraulic system incorporating the present invention, wherein indicated at 301 is a total hydrostatic support type hydraulic pump according to the invention using sea water as pressure medium, at 302 is a motor for driving the pump 301, at 303 is a strainer or filter, at 304 is a sea water pressure control valve, at 305 is a seat water-operated actuator, and at 306 is a controlling object which is driven by the actuator 305.

In this sea water hydraulic system arrangement, the actuator 305 is driven in the same manner as in an ordinary hydraulic system but the return operating fluid from the sea water pressure control valve 304, namely, used sea water is directly released into the sea.

When the hydraulic machine of the invention is applied as the total hydrostatic support type sea water hydraulic pump 301, the hydrostatic radial and thrust bearings are likewise supplied with its own discharge sea water pressure after modulations according to the tilt angle of the cylinder block, maintaining appropriate surface pressure on a hydrostatic sleeve as well as hydrostatic pads. It follows that, even with a low lubricative operating fluid like sea water, abnormal friction between sliding surfaces of the hydrostatic bearing and drive disc can be suitably prevented. Consequently, it becomes possible to provide a small-sized superhigh-pressure sea water pump with sufficient durability.

Although the invention has been described by way of total hydrostatic support type hydraulic pumps, it is to be understood that the invention can be realized as partial hydrostatic support type machines

employing the radial hydrostatic bearing in combination with a mechanical anti-friction bearing (e.g., a roll bearing) in place of the thrust hydrostatic bearing, or employing the hydrostatic thrust bearing in combination with a mechanical antifriction bearing in place of the hydrostatic radial bearing. In short, the present invention is applicable to hydraulic machines which include at least one of the hydrostatic radial and thrust bearings.

In a case where the hydraulic machine of the invention is applied as a reversible hydraulic motor, the paired suction and discharge ports formed in the valve plate as well as the paired suction and discharge passages formed in the head casing become a high pressure port. Therefore, the variable throttle mechanism needs to provide a pair of oil grooves or oil holes in each of the paired suction and discharge ports or passages, taking out a higher pressure by means of a shuttle valve as bearing control pressure for supply to the hydrostatic bearing.

Moreover, although the tilting mechanism 18 has been shown as being provided in the head casing 3 in the foregoing embodiments, it may be substituted by a tilting mechanism which is located on a side wall of the casing body 2 and which is arranged to tilt the cylinder block and valve plate through a yoke having one end thereof supported on a trunnion within the casing.

In addition to the examples of application given above, the hydraulic machine according to the invention is applicable to hydraulic systems of powder molding machines, injection molding machines, high speed forging machines operating in a high temperature enviroment, tunnel excavating machines and other hydraulically operated machines. Especially in case of an injection molding machine in which the dimensional accuracy of molded products is influenced by the control of hydraulic pressure, it becomes possible to enhance the accuracy of products by elevating the line pressure from the currently adopted level of about 14.7 MPa to 49 MPa for reducing fluctuations in the injecting pressure to 1/3 or less.

Even when applied to a hydraulic pump to be used under high pressure conditions, the bearing of the invention can support the rotational shaft in a stable manner.

POSSIBILITIES OF INDUSTRIAL APPLICATION

As described in detail hereinbefore, according to the present invention, the bearing control pressure corresponding to the tilt angle of the cylinder block is produced through a variable throttle mechanism and fed to at least a hydrostatic radial or thrust bearing to impart thereto a hydrostatic supporting capacity corresponding to the tilt angle of the cylinder block.

As a result, despite variations in radial or thrust loads of hydraulic reaction forces which are exerted on the drive disc through pistons, the positioning accuracy of the drive disc can be maintained constantly and the rotational shaft can be supported stably at any rotational speed.

The hydrostatic support of the drive disc according to the tilt angle of the cylinder block contributes to hold the leakage from the sliding surface between the drive disc and hydrostatic bearing to a minimum and constant rate, thereby minimizing power losses.

Further, since the hydrostatic bearing is supplied with a bearing control pressure varying according to the tilt angle of the cylinder block, it becomes possible to preclude abnormal frictional wear of the sliding guide surface of the hydrostatic bearing and to prevent deterioration in durability even if used under high pressure conditions for a long period of time, permitting continuous operations over long time periods.

Claims

1. A bent axis type variable displacement hydraulic machine, including: a cylindrical casing (1) having a head casing (3) with suction and discharge passages; a rotational shaft (5) rotatably inserted into said casing (1) and having a drive disc (6) at the distal end thereof disposed in said casing; a cylinder block (8) located in said casing and having a plural number of axial cylinder bores (9); a plural number of pistons (10) reciprocably received in said cylinder bores (9) in said cylinder block and each pivotally supported at one end by said drive disc (6); a valve plate (11) having a pair of suction and discharge ports (12, 13) and formed with a switching surface (11A) on one end face in sliding contact with said cylinder block (8) and a sliding surface (11B) on the other end face tiltably in sliding contact with a tilting slide surface on said cylinder block (8); a tilting mechanism (18) for tilting said valve plate (11) together with said cylinder block (8); at least one hydrostatic bearing (23; 28) provided between said drive disc (6) and said casing (1) as a radial or thrust bearing to support either radial or thrust load exerted on said drive disc (6) by hydraulic reaction forces; and throttle means (34, 38; 51, 54) for supplying hydraulic oil to the hydrostatic bearing (23; 28)
characterized in that

said a variable throttle means (34, 37; 51, 54) provided between said head casing (3) and said valve plate (11) are adapted to produce a pressure modulated in correspondence to the tilt angle of said cylinder block (8) for supply to said hydrostatic bearing (23; 28)

2. A bent axis type variable displacement hydraulic machine as defined in claim 1, wherein said hydrostatic bearing is a hydrostatic radial bearing (23), and said variable throttle means (34) comprises an oil groove (35) provided on said sliding surface (11B) of said valve plate (11) along and in communication with one of paired suction and discharge ports (12, 13) whichever is on the high pressure side and having a depth increasing with the tilt angle of said valve plate (11), and an oil hole (36) provided on said tilting slide surface (15) of said head casing (3) at a position confronting said oil groove (35), drawing out through said oil hole (36) a bearing control pressure increasing with the tilt angle of said valve plate (11) for supply to said hydrostatic radial bearing (23).

3. A bent axis type variable displacement hydraulic machine as defined in claim 1 or 2, wherein said hydrostatic bearing is a hydrostatic thrust bearing (28), and said variable throttle means (37) comprises an oil groove (38) provided on said sliding surface (11B) of said valve plate (11) along and in communication with one of paired suction and discharge ports (12, 13) whichever is on the high pressure side and having a depth decreasing correspondingly to increase in tilt angle of said valve plate (11), and an oil hole (39) provided on said tilting slide surface (15) of said head casing (3) at a position confronting said oil groove (38), drawing out through said oil hole (39) a bearing control pressure decreasing in inverse proportion to the tilt angle of said valve plate (11) for supply to said hydrostatic thrust bearing (28).

4. A bent axis type variable displacement hydraulic machine as defined in claim 2 or 3, wherein said oil groove (35, 38) formed on said valve plate (11) is communicated with said port on the higher pressure side at the deepest end thereof.

5. A bent axis type variable displacement hydraulic machine as defined in claim 1, wherein said hydrostatic bearing is a radial bearing (23), and said variable throttle means (51) comprises an oil groove (52) provided on said tilting slide surface (15) of said head casing (3) along and in communication with one of paired suction and discharge ports whichever is on the high pressure side and having a depth increasing with the tilt angle of said valve plate (11), and an oil hole (53) provided on said sliding surface (11B) of said valve plate (11) at a position confronting said oil groove (52) and in communication with one of paired suction and discharge ports whichever is on the high pressure side, drawing out through said oil groove (52) a bearing control pressure increasing with the tilt angle of said valve plate (11) for supply to said hydrostatic radial bearing (23).

6. A bent axis type variable displacement hydraulic machine as defined in claim 1, or 2 wherein said hydrostatic bearing is a thrust bearing (28), and said variable throttle means (54) comprises an oil groove (55) provided on said tilting slide surface (15) of said head casing (3) along and in communication with one of paired suction and discharge ports whichever is on the high pressure side and having a depth decreasing correspondingly to increase in tilt angle of said valve plate (11), and an oil hole (56) provided on said sliding surface (11B) of said valve plate (11) at a position confronting said oil groove (55) and in communication with one of paird suction and discharge ports whichever is on the high pressure side, drawing out through said oil groove (55) a bearing control pressure decreasing in inverse proportion to the tilt angle of said valve plate (11) for supply to said hydrostatic thrust bearing (28).

7. A bent axis type variable displacement hydraulic machine as defined in claim 5 or 6, wherein said oil groove (55) formed on said head casing (3) is provided with an oil passage at the deepest end thereof for drawing out said bearing control pressure.

8. A bent axis type variable displacement hydraulic machine, including: a cylindrical casing (1) having a head casing (3) with suction and discharge passages; a rotational shaft (5) rotatably inserted into said casing and having a drive disc (6) at the distal end thereof disposed in said casing; a cylinder block (8) located in said casing and having a plural number of axial cylinder bores (9); a plural number of pistons (10) reciprocably received in said cylinder bores (9) in said cylinder block (8) and each pivotally supported at one end by said drive disc (6); a valve plate (11) having a pair of suction and discharge ports and formed with a switching surface (11A) on one end face in sliding contact with said cylinder

block (8) and a sliding surface (11B) on the other end face tiltably in sliding contact with a tilting slide surface on said cylinder block (8); a tilting mechanism (18) for tilting said valve plate (11) together with said cylinder block (8); and at least one hydrostatic bearing (23; 28) provided between said drive disc (6) and said casing (1) as a radial or thrust bearing to support either radial or thrust load exerted on said drive disc (6) by hydraulic reaction forces;
characterized by
a sensor (71) for detecting the tilt angle of said cylinder block (8) inclined by said tilting mechanism (18);
an oil passage (72; 73) for drawing out pressure from one of said paired suction and discharge passages whichever is on the high pressure side, for supply to said hydrostatic bearing (23; 28); and
a control valve (77; 78) located within the length of said oil passage and adapted to modulate the pressure according to a tilt angle signal received from said sensor (71).

9. A bent axis type variable displacement hydraulic machine as defined in claim 8, wherein said hydrostatic bearing is a radial bearing (23), and said control valve is an electromagnetic proportional valve (77) producing a bearing control pressure increasing with said tilt angle.

10. A bent axis type variable displacement hydraulic machine as defined in claim 8 or 9, wherein said hydrostatic bearing is a thrust bearing (28), and said control valve is an electromagnetic proportional reducing valve (78) producing a bearing control pressure lowering inversely to said tilt angle.

11. A bent axis type variable displacement hydraulic machine as defined in claim 1 or 8, wherein said hydraulic machine is applied as a pump of main hydraulic pressure source or a drive motor in a hydraulic system of construction machine.

12. A bent axis type variable displacement hydraulic machine as defined in claim 1 or 8, wherein said hydraulic machine is applied as a main pump in a hydraulic system for a screw down mechanism of rolling mill.

13. A bent axis type variable displacement hydraulic machine as defined in claim 1 or 8, wherein said hydraulic machine is applied as a pump of main hydraulic pressure source in a sea water hydraulic system.

Patentansprüche

1. Hydraulische verstellmaschine vom Schrägachsentyp, die umfaßt: ein zylindrisches Gehäuse (1) mit einem Kopfgehäuse (3), mit Ansaug- und Auslaßkanälen, eine Drehwelle (5), die drehbar in das Gehäuse (1) eingesetzt ist und am distalen Ende eine Antriebsscheibe (6) aufweist, die im Gehäuse (1) angeordnet ist, einen im Gehäuse angeordneten Zylinderblock (8), der mehrere axiale Zylinderbohrungen (9) aufweist, mehrere Kolben (10), deren hin- und hergehende Bewegung in den Zylinderbohrungen (9) im Zylinderblock aufgenommen wird, und wobei jeder Kolben an einem Ende durch die Antriebsscheibe (6) drehbar gelagert ist, eine Ventilplatte (11), die ein Paar Ansaug- und Auslaßschlitze (12, 13) aufweist, und bei der eine Schaltfläche (11A) an einer Endfläche in Gleitkontakt mit dem Zylinderblock (8) ausgebildet ist, und die andere Endfläche kippbar im Gleitkontakt mit einer Neigungsgleitfläche des Zylinderblocks (8) steht, ein Kippmechanismus (18) zum Kippen der Ventilplatte (11) zusammen mit dem Zylinderblock (8), mindestens ein hydrostatisches Lager (23; 28) zwischen der Antriebsscheibe (6) und dem Gehäuse (1) als ein Radiallager oder Axiallager, um entweder die Radiallast oder die Axiallast abzustützen, die durch hydraulische Reaktionskräfte auf die Antriebsscheibe (6) ausgeübt wird, und eine Drosselvorrichtung (34, 38; 51, 54) zur Zuleitung von Hydrauliköl zum hydrostatischen Lager (23; 28),
dadurch gekennzeichnet,
daß die regelbaren Drosselvorrichtungen (34, 38; 51, 54) zwischen dem Kopfgehäuse (3) und der Ventilplatte (11) derart ausgelegt sind, daß sie einen Druck erzeugen, der gemäß dem Kippwinkel des Zylinderblocks (8) zur Zuleitung zum hydrostatischen Lager (23; 28) geregelt wird.

2. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1, wobei das hydrostatische Lager ein hydrostatisches Radiallager (23) ist, und die regelbare Drosselvorrichtung (34) eine Ölnut (35) umfaßt, die auf der Gleitfläche (11B) der Ventilplatte (11) entlang und in Verbindung mit einem Schlitz

von dem Paar der Ansaug- und Auslaßschlitze (12, 13) angeordnet ist, der sich auf der Hochdruckseite befindet, und eine Tiefe aufweist, die mit dem Anstieg des Kippwinkels der Ventilplatte (11) zunimmt, und ein Ölloch (36), das auf der Neigungsgleitfläche (15) des Kopfgehäuses (3) in einer Stellung angeordnet ist, die der Ölnut (35) gegenüberliegt, und das zur Zufuhr zum hydrostatischen Radiallager (23) durch das Ölloch (36) einen Lagersteuerdruck entnimmt, der mit dem Kippwinkel der Ventilplatte (11) zunimmt.

3. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1 oder 2, wobei das hydrostatische Lager ein hydrostatisches Axiallager (28) ist, und die regelbare Drosselvorrichtung (37) eine Ölnut (38) umfaßt, die auf der Gleitfläche (11B) der Ventilplatte (11) entlang und in Verbindung mit einem Schlitz von dem Paar der Ansaug- und Auslaßschlitze (12, 13) angeordnet ist, der sich auf der Hochdruckseite befindet, und eine Tiefe aufweist, die mit dem Anstieg des Kippwinkels der Ventilplatte (11) abnimmt, und ein Ölloch (39), das auf der Neigungsgleitfläche (15) des Kopfgehäuses (3) in einer Stellung angeordnet ist, die der Ölnut (38) gegenüberliegt, und das zur Zufuhr zum hydrostatischen Axiallager (28) durch das Ölloch (39) einen Lagersteuerdruck entnimmt, der umgekehrt proportional zum Kippwinkel der Ventilplatte (11) abnimmt.

4. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 2 oder 3, wobei die auf der Ventilplatte (11) ausgebildete Ölnut (35, 38) auf der Seite mit dem höheren Druck an ihrem tiefsten Ende mit dem Schlitz in Verbindung steht.

5. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1, wobei das hydrostatische Lager ein hydrostatisches Radiallager (23) ist, und die regelbare Drosselvorrichtung (51) eine Ölnut (52) umfaßt, die auf der Gleitfläche (15) des Kopfgehäuses (3) entlang und in Verbindung mit einem Schlitz vom Paar der Ansaug- und Auslaßschlitze angeordnet ist, der sich auf der Hochdruckseite befindet, und eine Tiefe aufweist, die mit dem Kippwinkel der Ventilplatte (11) zunimmt, und ein Ölloch (53), das auf der Gleitfläche (11B) der Ventilplatte (11) in einer Stellung angeordnet ist, die der Ölnut (52) gegenüberliegt, und mit einem Schlitz des Paars der Ansaug- und Auslaßschlitze in Verbindung steht, der sich auf der Hochdruckseite befindet, und zur Zufuhr zum hydrostatischen Radiallager (23) durch die Ölnut (52) einen Lagersteuerdruck entnimmt, der mit dem Kippwinkel der Ventilplatte (11) zunimmt.

6. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1 oder 2, wobei das hydrostatische Lager ein hydrostatisches Axiallager (28) ist, und die regelbare Drosselvorrichtung (54) eine Ölnut (55) umfaßt, die auf der Neigungsgleitfläche (15) des Kopfgehäuses (3) entlang und in Verbindung mit einem Schlitz des Paars der Ansaug- und Auslaßschlitze angeordnet ist, der sich auf der Hochdruckseite befindet, und eine Tiefe aufweist, die mit dem Zunehmen des Kippwinkels der Ventilplatte (11) abnimmt, und ein Ölloch (56), das auf der Gleitfläche (11B) der Ventilplatte (11) in einer Stellung angeordnet ist, die der Ölnut (55) gegenüberliegt, und mit einem Schlitz des Paars der Ansaug- und Auslaßschlitze in Verbindung steht, der sich auf der Hochdruckseite befindet, und zur Zufuhr zum hydrostatischen Axiallager (28) durch die Ölnut (55) einen Lagersteuerdruck entnimmt, der umgekehrt proportional zum Kippwinkel der Ventilplatte (11) abnimmt.

7. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 5 oder 6, wobei die auf dem Kopfgehäuse (3) ausgebildete Ölnut (55) an ihrem tiefsten Ende mit einem Ölkanal zum Entnehmen des Lagersteuerdrucks ausgestattet ist.

8. Hydraulische Verstellmaschine vom Schrägachsentyp, die umfaßt: ein zylindrisches Gehäuse (1) mit einem Kopfgehäuse (3), mit Ansaug- und Auslaßkanälen, eine Drehwelle (5), die drehbar in das Gehäuse (1) eingesetzt ist, und am distalen Ende eine Antriebsscheibe (6) aufweist, die im Gehäuse (1) angeordnet ist, einen im Gehäuse angeordneten Zylinderblock (8), der mehrere axiale Zylinderbohrungen (9) aufweist, mehrere Kolben (10), deren hin- und hergehende Bewegung in den Zylinderbohrungen (9) im Zylinderblock (8) aufgenommen wird, und wobei jeder Kolben an einem Ende durch die Antriebsscheibe (6) drehbar gelagert ist, eine Ventilplatte (11), die ein Paar Ansaug- und Auslaßschlitze aufweist, und bei der eine Schaltfläche (11A) an einer Endfläche in Gleitkontakt mit dem Zylinderblock (8) ausgebildet ist, und eine Gleitfläche (11B) auf der anderen Endfläche kippbar im Gleitkontakt mit einer Neigungsgleitfläche auf dem Zylinderblock (8) steht, ein Kippmechanismus (18) zum Kippen der Ventilplatte (11) zusammen mit dem Zylinderblock (8), mindestens ein hydrostatisches Lager (23; 28) zwischen der Antriebsscheibe (6) und dem Gehäuse (1) als ein Radiallager oder Axiallager, um

entweder die Radiallast oder die Axiallast abzustützen, die durch hydraulische Reaktionskräfte auf die Antriebsscheibe (6) ausgeübt wird,

gekennzeichnet durch

- einen Sensor (71) zum Erfassen des Kippwinkels des Zylinderblocks (8), der durch den Kippmechanismus (18) gekippt wird,
- einen Ölkanal (72; 73) der zur Zufuhr zum hydrostatischen Lager (23; 28) von einem Schlitz des Paars der Ansaug- und Auslaßschlitze, der sich auf der Hochdruckseite befindet, Druck entnimmt und
- ein in der Länge des Ölkanals angeordnetes Steuerventil (77; 78), das zum Regeln bzw. Umwandeln des Drucks gemäß einem vom Sensor (71) erhaltenen Kippwinkelsignal ausgelegt ist.

9. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 8, wobei das hydrostatische Lager ein Radiallager (23) und das Steuerventil ein elektromagnetisches Proportionalventil (77) ist, das einen Lagersteuerdruck erzeugt, der mit dem Kippwinkel zunimmt.

10. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 8 oder 9, wobei das hydrostatische Lager ein Axiallager (28) und das Steuerventil ein elektromagnetisches Proportionalreduzierventil (78) ist, das einen Lagersteuerdruck erzeugt, der abnimmt, wenn der Kippwinkel zunimmt.

11. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1 oder 8, wobei die hydraulische Maschine als Pumpe, die von einer Hauptquelle mit einem hydraulischen Druck versorgt wird, oder als Antriebsmotor in einem Hydrauliksystem einer Baumaschine verwendet wird.

12. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1 oder 8, wobei die Hydraulikmaschine als Hauptpumpe in einem Hydrauliksystem für einen Anziehmechanismus eines Walzwerks verwendet wird.

13. Hydraulische Verstellmaschine vom Schrägachsentyp gemäß Anspruch 1 oder 8, wobei die Hydraulikmaschine als Pumpe einer Haupthydraulikquelle in einem Seewasser-Hydrauliksystem verwendet wird.

Revendications

1. Machine hydraulique à cylindrée variable du type à axe brisé, comprenant: un carter cylindrique (1) comportant un carter de tête (3) muni de passages d'aspiration et de refoulement; un arbre rotatif (5) inséré de façon tournante dans ledit carter (1) et comportant un disque d'entraînement (6) à son extrémité distale disposée dans ledit carter; un bloc cylindre (8) placé dans ledit carter et comportant plusieurs alésages axiaux cylindriques (9); plusieurs pistons (10) logés en vue d'un mouvement de va-et-vient dans lesdits alésages cylindriques (9) du bloc cylindre et supportés chacun de façon pivotante à une de leurs extrémités par le disque d'entraînement (6); un plateau-soupape (11) comportant une paire d'orifices d'aspiration et de refoulement (12, 13) et pourvu d'une surface de commutation (11A) sur une de ses faces d'extrémité en contact glissant avec le bloc cylindre (8) et une surface de glissement (11B) sur l'autre face d'extrémité en contact glissant, de façon inclinable, avec une surface glissante d'inclinaison du bloc cylindre (8); un mécanisme d'inclinaison (18) pour incliner le plateau-soupape (11) conjointement avec le bloc cylindre (8); au moins un palier hydrostatique (23, 28) disposé entre le disque d'entraînement (6) et le carter (1) en tant que palier radial ou palier de butée pour supporter soit une charge radiale, soit une charge de poussée exercée sur le disque d'entraînement (6) par des forces de réaction hydrauliques; et des moyens d'admission variable (34, 37; 51, 54) pour alimenter en huile hydraulique le palier hydrostatique (23; 28),

   caractérisée en ce que:

   lesdits moyens d'admission variable (34, 37; 51, 54) disposés entre le carter de tête (3) et le plateau-soupape (11) sont adaptés pour produire une pression qui est modulée en correspondance avec l'angle d'inclinaison du bloc culindre (8) pour alimenter ledit palier hydrostatique (23; 28).

2. Machine hydraulique à cylindrèe variable du type à axe brisé selon la revendication 1, dans laquelle le palier hydrostatique est un palier hydrostatique radial (23) et les moyens d'admission variables (34) comprennent une rainure (35) à huile formée sur ladite surface glissante (11B) du plateau-soupape (11), le long d'un des orifices d'aspiration et de refoulement appariés (12, 13) et en communication avec cet orifice, qu'il se trouve ou non sur le côté haute pression, et ayant une profondeur qui

augmente avec l'angle d'inclinaison du plateau-soupape (11), et un trou (36) à huile formé sur la surface d'inclinaison glissante (15) du carter de tête (3) à un endroit qui se trouve en regard de la rainure (35) à huile, une pression de commande de palier qui augmente avec l'angle d'inclinaison du plateau-soupape (11) pour alimenter le palier hydrostatique radial (23).

3. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 1 ou 2, dans laquelle ledit palier hydrostatique est un palier de butée hydrostatique (28) et lesdits moyens d'admission variables (37) comprennent une rainure (38) à huile formée dans ladite surface glissante (11B) du plateau-soupape (11), le long d'un des orifices d'aspiration et de refoulement appariés (12, 13) et en communication avec cet orifice, qu'il se trouve ou non sur le côté haute pression, et ayant une profondeur qui diminue de façon correspondante à l'augmentation de l'angle d'inclinaison du plateau-soupape (11), et un trou (39) à huile formé dans ladite surface glissante d'inclinaison (15) du carter de tête (3) à un endroit situé en regard de la rainure (38) à huile, une pression de commande de palier qui diminue en raison inverse de l'angle d'inclinaison du plateau de soupapes (11) pour alimenter le palier de butée hydrostatique (28) étant prélevée, par l'intermédiaire dudit trou (36) à huile.

4. Machine hydraulique à cylindrèe variable du type à axe brisé selon la revendication 2 ou 3, dans laquelle ladite rainure (35, 38) à huile formée sur le plateau-soupape (11) communique avec ledit orifice sur le côtè de pression plus élevée à l'extrémité la plus profonde de cette rainure.

5. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 1, dans laquelle le palier hydrostatique est un palier radial (23) et le moyen d'admission variable (51) comprend une rainure (52) à huile formée dans ladite surface glissante d'inclinaison (15) du carter de tète (3), le long d'un des orifices d'aspiration et de refoulement et en communication avec cet orifice, qu'il se trouve ou non sur le côté haute pression, et ayant une profondeur qui augmente avec l'angle d'inclinaison du plateau-soupape (11), et un trou (53) à huile formé sur la surface glissante (11B) du plateau-soupape (11) à un endroit situé en regard de la rainure (52) à huile et en communication avec un des orifices d'aspiration et de refoulement apporiés, qu'il se trouve ou non sur le côté haute pression, une pression de commande de palier qui augmente avec l'angle d'inclinaison du plateau-soupape (11) pour alimenter le palier hydrostatique radial (23) étant prélevée par l'intermédiaire de la rainure (52) à huile.

6. Machine hydraulique à cylindrèe variable du type à axe brisé selon la revendication 1 ou 2, dans laquelle ledit palier hydrostatique est un palier de butée (28), et le moyen d'admission variable (54) comprend une rainure (55) à huile formée dans ladite surface glissante d'inclinaison (15) du carter de tête (3), le long d'un des orifices d'aspiration et de refoulement appariés et en communication avec cet orifice, qu'il se trouve ou non sur le côté haute pression et ayant une profondeur qui diminue de façon correspondante à l'augmentation de l'angle d'inclinaison du plateau-soupape (11), et un trou (56) à huile formé sur la surface glissante (11B) du plateau-soupape (11) à un endroit situé en regard de la rainure (55) à huile et en communication avec un des orifices d'aspiration et de refoulement appariés, qu'il se trouve ou non sur le côté haute pression, une pression de commande de palier qui diminue en raison inverse de l'angle d'inclinaison du plateau-soupape (11) pour alimenter le palier de butée hydrostatique (28) étant prélevée par l'intermédiaire de la rainure (55) à huile.

7. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 5 ou 6, dans laquelle la rainure (55) à huile formée sur le carter de tête (3) est pourvue d'un passage d'huile à son extrémité la plus profonde pour appliquer ladite pression de commande de palier.

8. Machine hydraulique à cylindrée variable du type à axe brisé, comprenant: un carter cylindrique (1) comportant un carter de tête (3) pourvu de passages d'aspiration et de refoulement; un arbre rotatif (5) inséré de façon tournante dans ledit carter et comportant un disque d'entraînement (6) à son extrémité distale disposée dans ledit carter; un bloc cylindre (8) placé dans ledit carter et comportant plusieurs alésages cylindriques axiaux (9); plusieurs pistons (10) logés en vue d'un mouvement de va-et-vient dans lesdits alésages cylindriques (9) du bloc cylindre (8) et chacun étant supporté de façon pivotante à une de ses extrémités par le disque d'entraînement (6); un plateau-soupape (11) comportant une paire d'orifices d'aspiration et de refoulement et pourvu d'une surface de commutation (11A) sur une de ses faces d'extrémité en contact glissant avec le bloc cylindre (8) et une surface glissante (11B) sur son autre face en contact glissant, de façon inclinable, avec une surface glissante d'inclinaison se trouvant sur le bloc cylindre (8); un mécanisme d'inclinaison (18) pour incliner le plateau-soupape (11)

conjointement avec le bloc cylindre (8); et au moins un palier hydrostatique (23; 28) disposé entre le disque d'entraînement (6) et le carter (1) en tant que palier radial ou palier de butée pour supporter soit une charge radiale soit une charge de poussée exercée sur le disque d'entraînement (6) par des forces de réaction hydrauliques;

caractérisée par:

un capteur (71) pour détecter l'angle d'inclinaison du bloc cylindre (8) incliné par le wécanisme d'inclinaison (18);

un passage (72; 73) á huile pour prélever la pression d'un des passages d'aspiration et de refoulement appariés, qu'il se trouve ou non sur le côté haute pression, afin d'alimenter le palier hydrostatique (23; 28); et

une valve de commande (77; 78) placée dans les limites de la longueur dudit passage à huile et adaptée pour moduler la pression en fonction d'un signal d'angle d'inclinaison reçu du capteur (71).

9. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 8, dans laquelle le palier hydrostatique est un palier radial (23), et la valve de commande est une électrovalve proportionnelle (77) produisant une pression de commande de palier qui augmente avec ledit angle d'inclinaison.

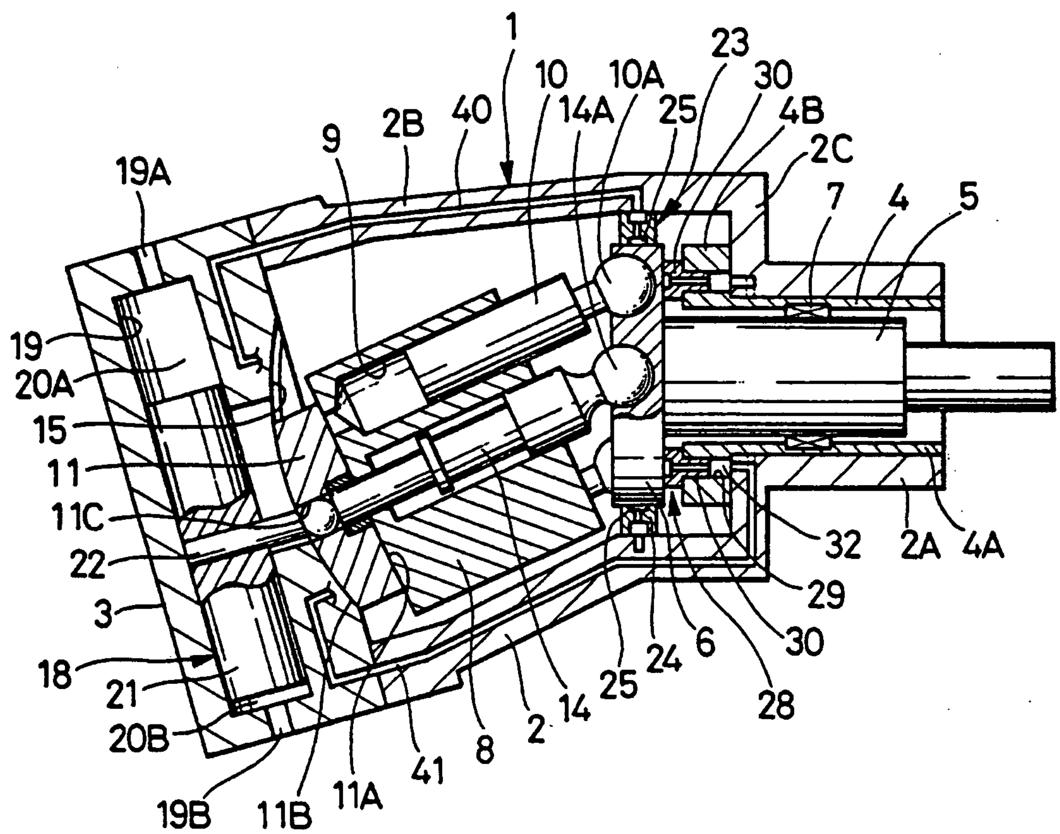
10. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 8 ou 9, dans laquelle ledit palier hydrostatique est un palier de butée (28) et la valve de commande est une électrovalve réductrice proportionnelle (78) produisant une pression de commande de palier diminuant en raison inverse dudit angle d'inclinaison.

11. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 1 ou 8, ladite machine hydraulique étant utilisée comme une pompe servant de source de pression hydraulique principale ou comme un moteur dans un système hydraulique d'engins de travaux.

12. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 1 ou 8, ladite machine hydraulique étant utilisée comme une pompe principale dans un système hydraulique pour un mécanisme de réglage de laminoir.

13. Machine hydraulique à cylindrée variable du type à axe brisé selon la revendication 1 ou 8, ladite machine étant utilisée comme une pompe servant de source de pression hydraulique principale dans un système hydraulique à eau de mer.

Fig. 1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2B
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4B
9
2C
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20A
15
11
11C
22
3
18
21
20B
19B
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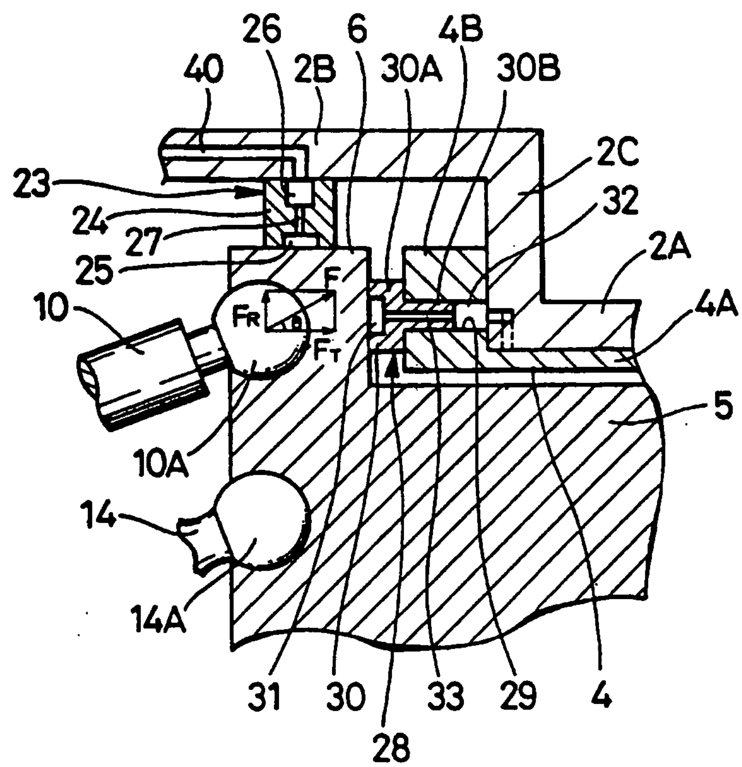
2A
4A

EP 0 389 644 B1

Fig. 2



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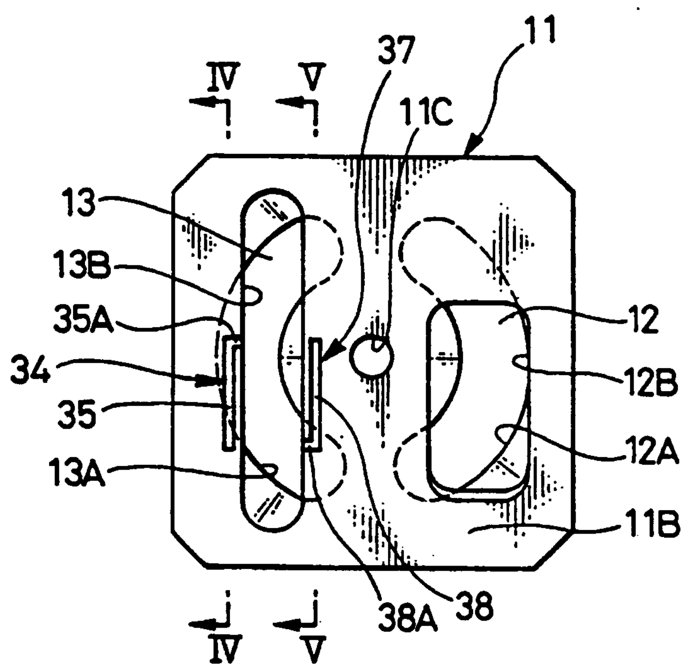

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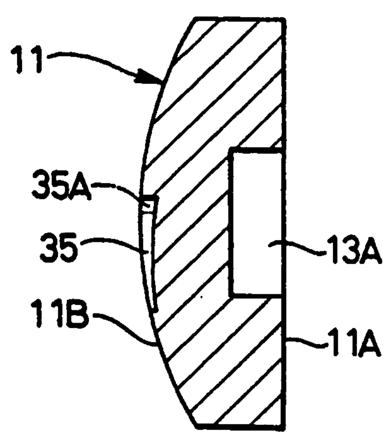

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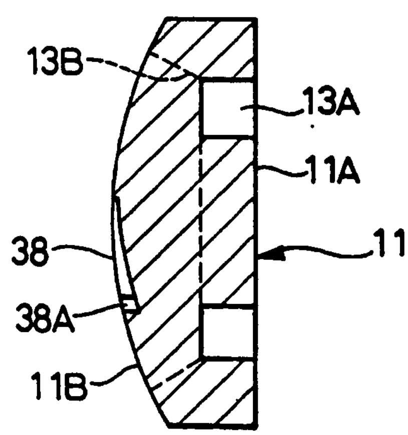

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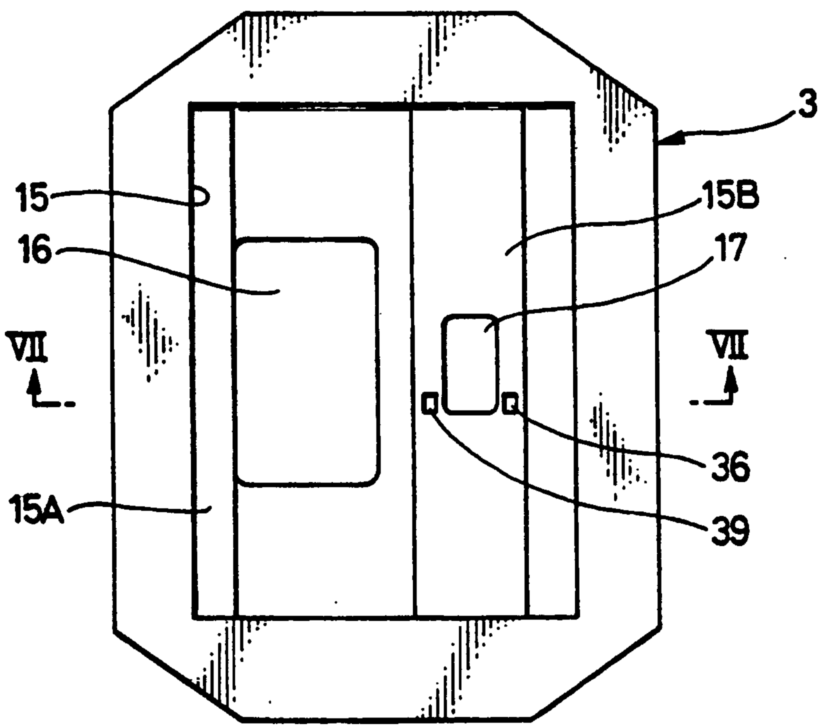

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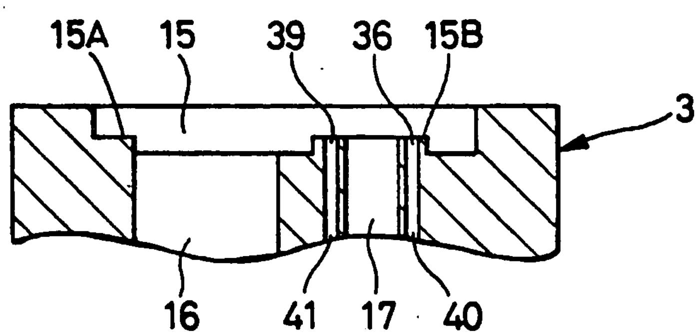
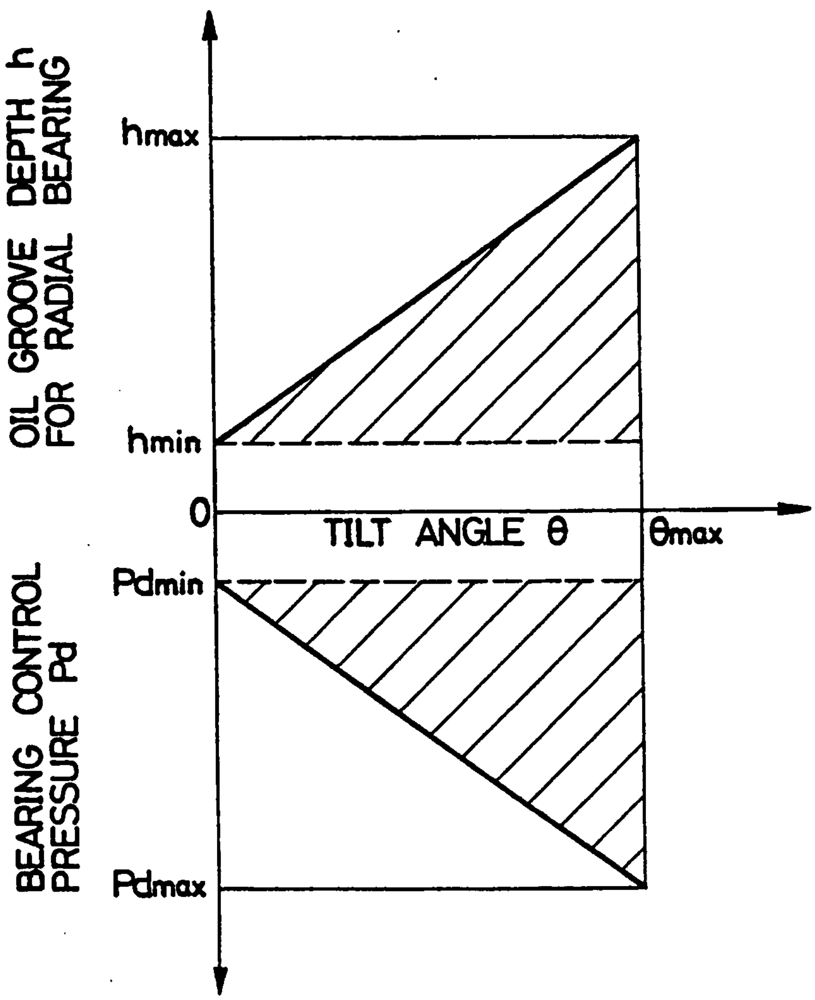


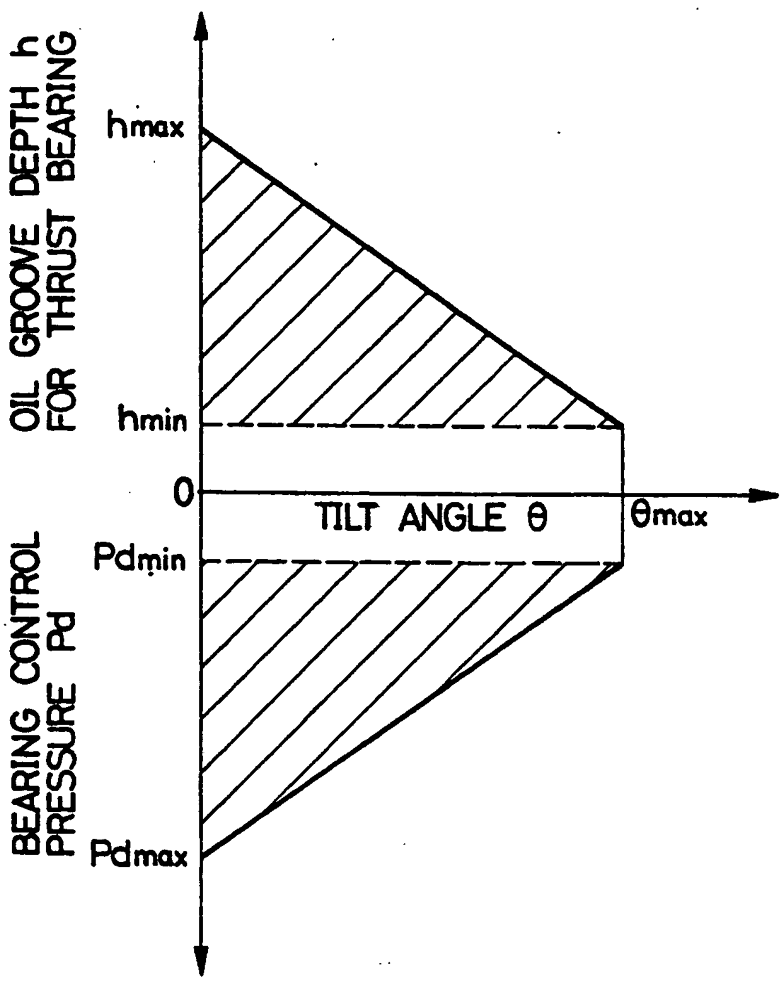
EP 0 389 644 B1

Fig. 8



EP 0 389 644 B1

Fig. 9



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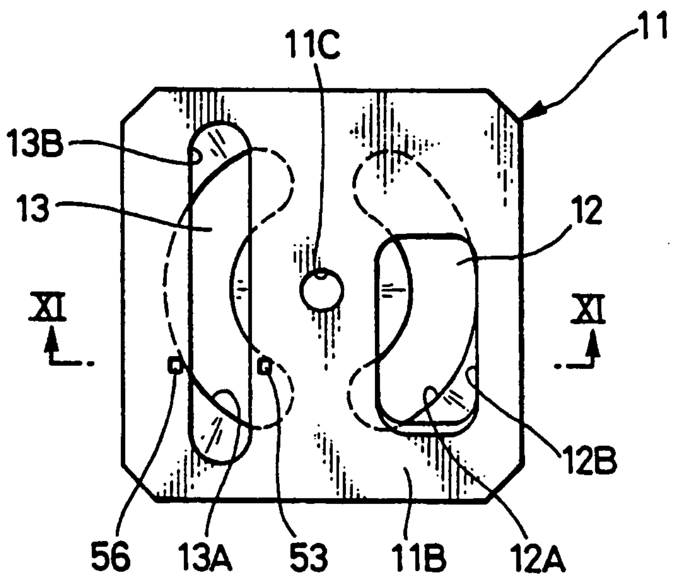
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11C
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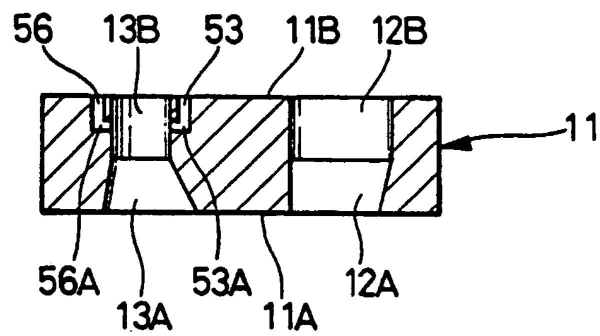
13
12
XI
XI
12B
56
13A
53
11B
12A

Fig. 11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12A

EP 0 389 644 B1

Fig. 12

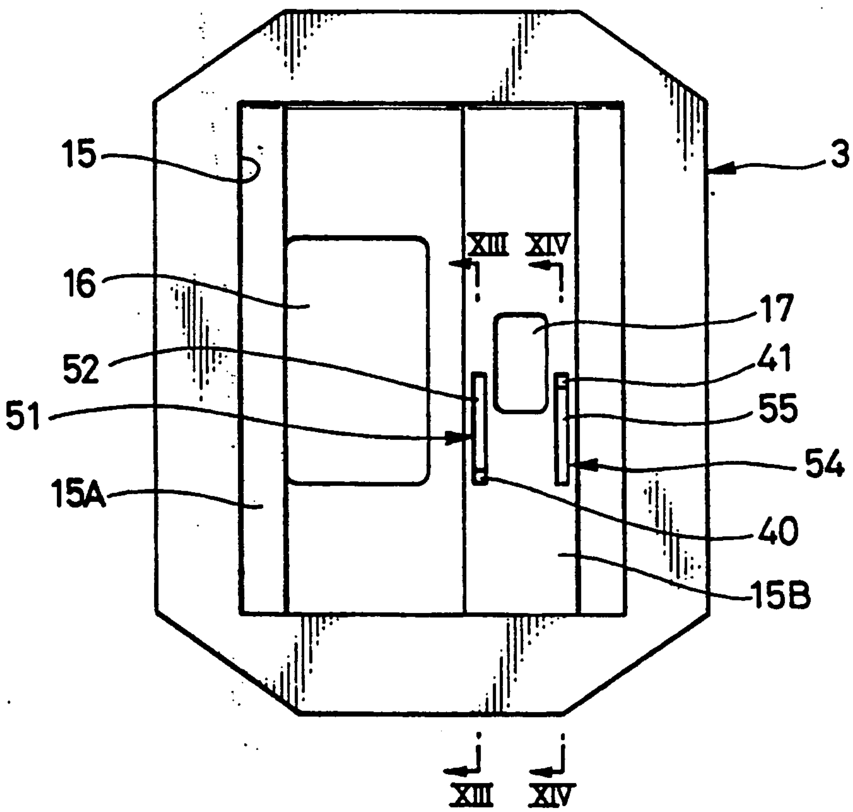


Fig. 13

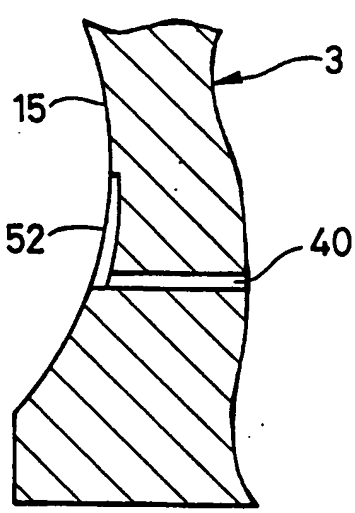
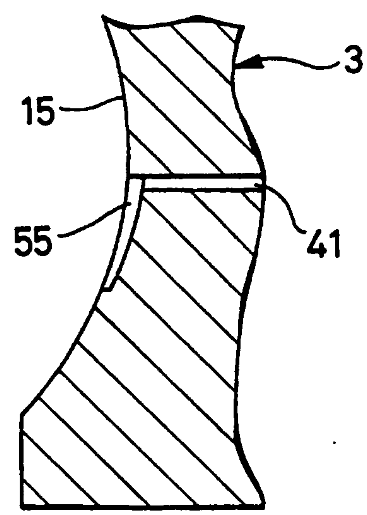
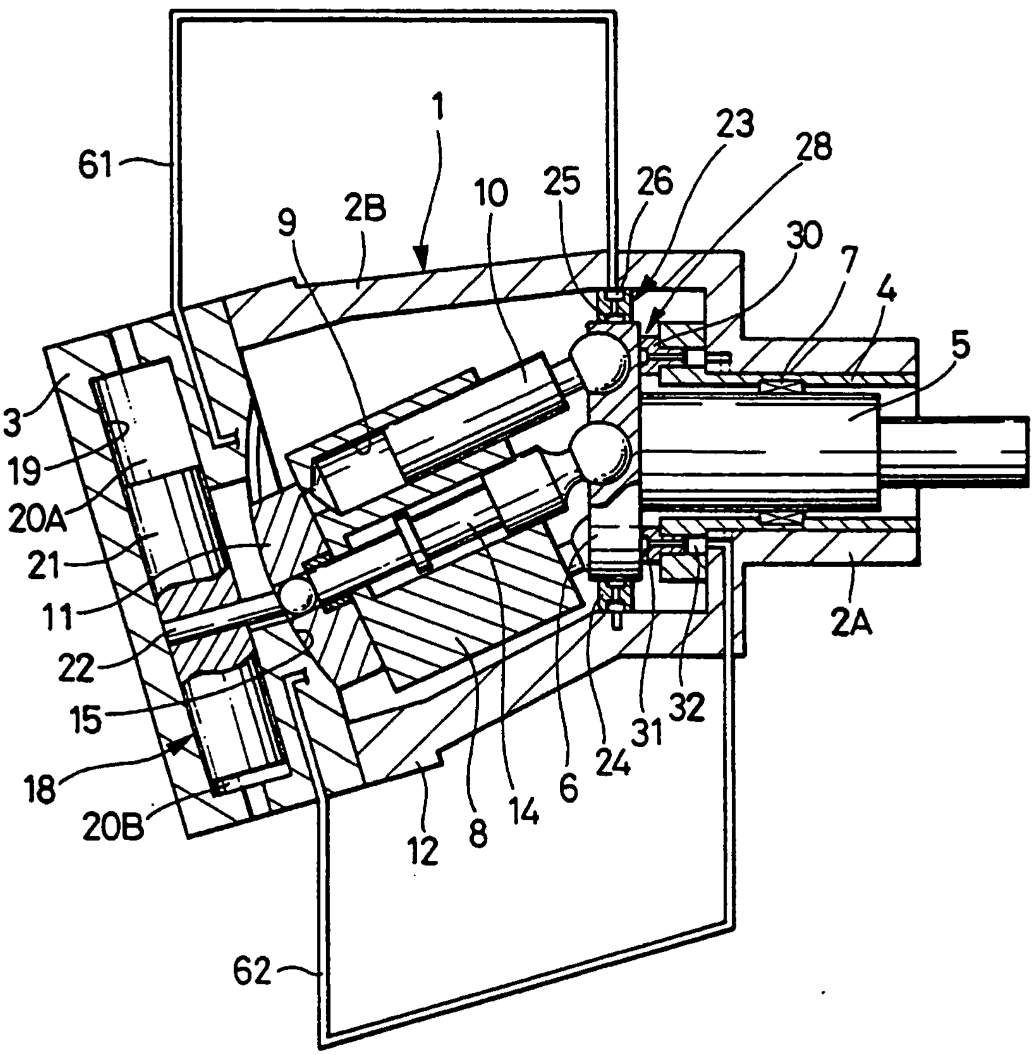


Fig. 14



EP 0 389 644 B1

Fig. 15



EP 0 389 644 B1

Fig. 16

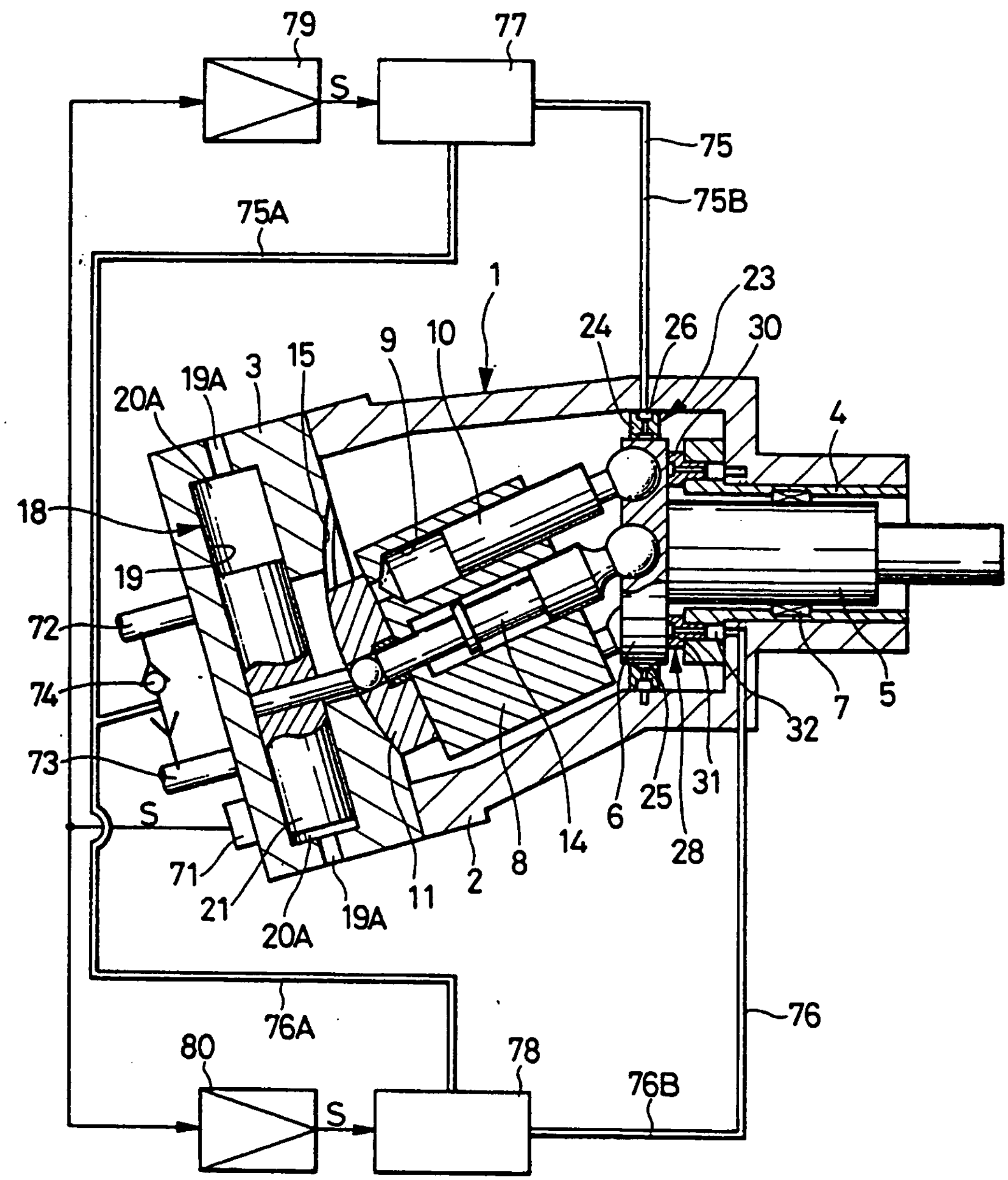


Fig. 17

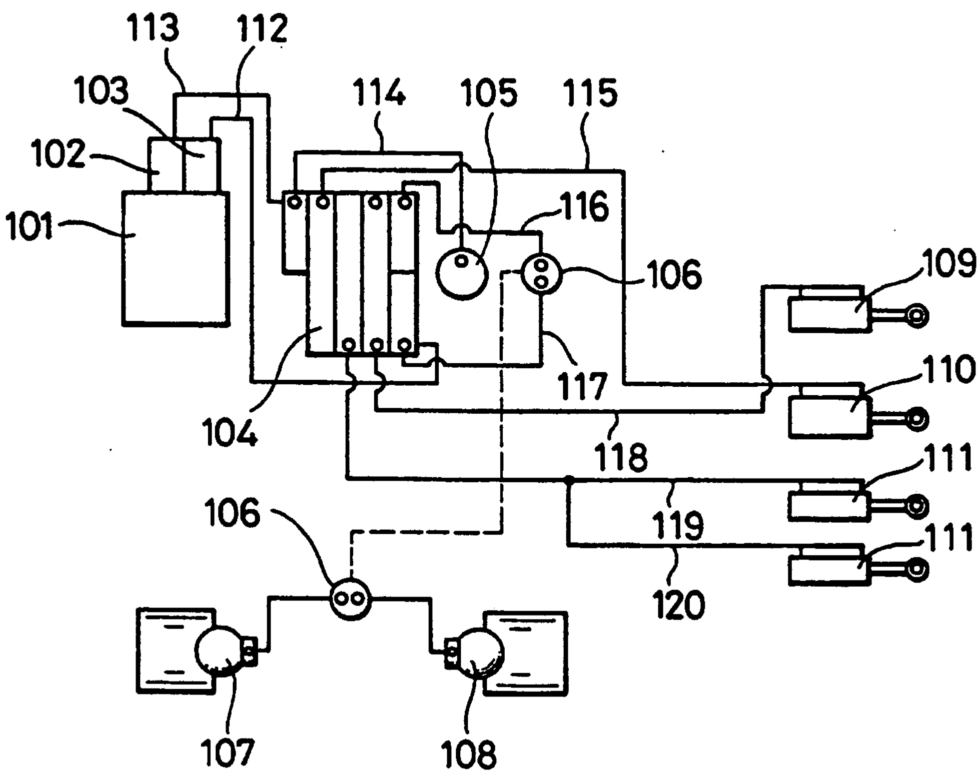

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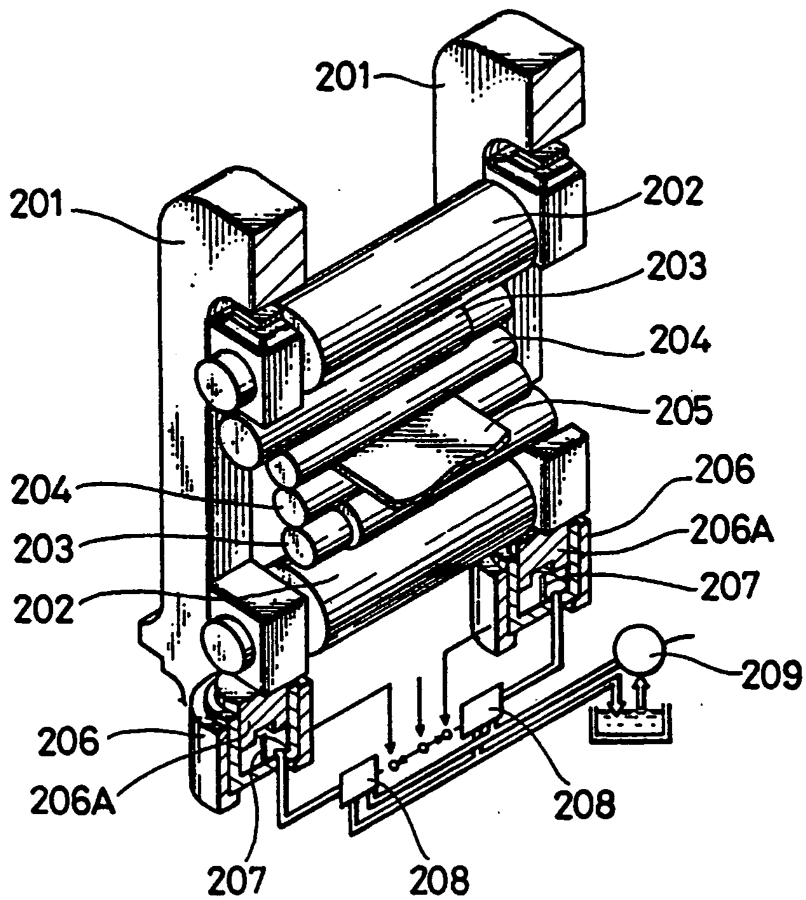
Fig. 18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206A
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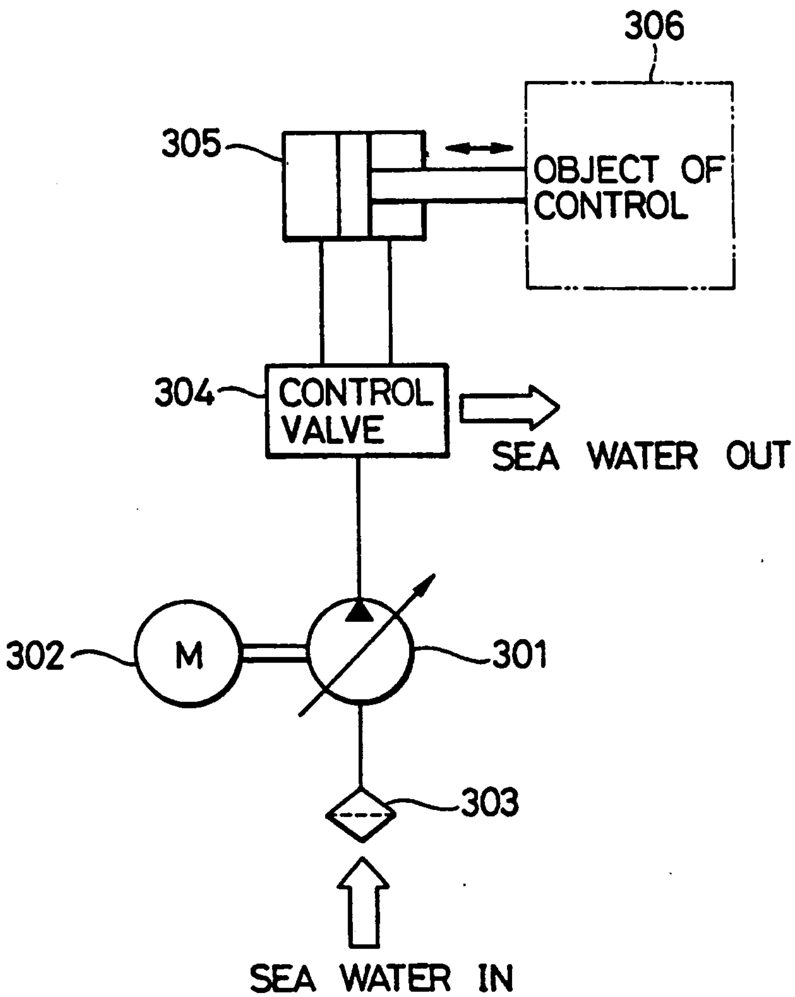
Fig. 19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OBJECT OF CONTROL
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CONTROL VALVE
SEA WATER OUT
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M
301
303
SEA WATER IN